John Laub -- JPL

Development of Elastic Orifices for Gas Bearings

The Gas Bearing Project at JPL

In the fall of 1957 I joined the Guidance and Control Division of the Jet Propulsion Laboratory (JPL) of the California Institute of Technology in Pasadena as a Research Specialist.

JPL at the time was under contract by the US Army Ordnance Department, and its assignments included the development of medium range tactical missiles of the Corporal and Sergeant type. The Corporal missile used a radio guidance system while the follow-on Sergeant was guided to the target by an all-inertial system consisting of a three axis gyro controlled platform on which the sensors, accelerometers, etc., and associated circuits were mounted.

JPL had been using a type of liquid floated gyro developed by the Instrumentation Laboratory of MIT. In this gyro the spinning wheel is encased in a cylindrical metal can which floats by buoyancy in a special liquid. This greatly reduces the load on the gimbal axis bearings and increases the sensitivity and accuracy of the gyro. JPL's experience with this type of gyro had been generally quite good. However, the temperature of the flotation fluid had to be closely controlled by electric heating since the viscosity and density of the fluid depend on its temperature. Difficulties were also experienced occasionally from leakage of the liquid and the effect of high-g accelerations and vibrations in military missiles.

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JPL maintained close liaison with the Redstone Arsenal of the US Army in Huntsville, Alabama through mutual visits of key people and otherwise. Mr. William Merrick who was then a Group Leader in the Guidance and Control Division, on one of his visits to Redstone Arsenal had become familiar with a gas floated type of gyro. It was originally developed by the German rocket laboratories in Peenemünde during the second world war and was used in the Redstone and Jupiter missiles by the team of German rocket experts, under Dr. Wernher von Braun, whom the army had brought over from Germany after the war. Redstone Arsenal adopted the Peenemünde gyro with only slight modifications. It was used in a considerable number of missiles with good success; however, it was expensive to manufacture and great care had to be exercised to eliminate unwanted turbine torques. The Guidance and Control Section of Redstone Arsenal had never undertaken a thorough study of the gas floated gyro with the aim of gaining a better understanding of the performance and optimizing it.

Mr. Merrick realized the inherent advantages of the gas flotation over liquid flotation, i.e., much lower viscosity and therefore friction, insensitivity to temperature, elimination of the leakage of the liquid, etc. He felt that JPL should become more familiar with gas floated gyros and should evalute their usefulness for JPL's guidance systems in comparison with liquid floated gyros. He arranged for one of Redstone's gyros to be shipped to JPL; they had expected to add a person qualified to work on it to their staff when I first met Mr. Merrick. He explained the problem and the need for a thorough investigation to me and I became sufficiently intrigued to agree to undertake the project. I had beome very interested in fluid dynamics in the course of the development of a boundary layer type mass flowmeter (see case history ECL-33) and felt that with the support available at JPL my venture into this "terra incognita" was perhaps reckless but promised to be exciting and, hopefully, rewarding. I had also been told by Mr. Merrick and others that a very venturesome attitude prevailed at JPL and nobody was afraid of failures from which one could always learn something of interest or usefulness.

Investigators at JPL had available excellent facilities and were supported by a number of special departments which relieved them from many chores that could be done much better by experts in their field. Some of these are shown on the chart, Exhibit I, which reflects the organization (in part) which existed at JPL at the time I joined it. Of particular interest to the kind of project I was about to embark on were a large computer section with several analog and digital computers, a large library with a special staff of literature searchers and a design and engineering section whose personnel was available for the detailed drawing and designing of apparatus and test fixtures from rough hand sketches. I was also impressed by the wide scope of activities and the unique mixture of academic and industrial atmosphere at JPL.

The basic design of the gas bearing of the Redstone Arsenal gyro can be seen in Exhibit II. A cylindrical shell of 2 16 inch diameter and 2.20 inch length which houses the gyro wheel is floated in compressed gas, both radially and axially, and pivots around the output axis with virtually zero friction. The gas is fed to the radial and axial gap of 0.016 inch average width through orifices which automatically regulate the pressure in the gap under varying load or acceleration. The gas enters the gap through four rows of radial orifices and through a number of axial orifices in the thrust end plates and leaves through the clearance between the plates and the cylindrical studs which are attached to the shell. The rotation of the shell is restricted to a few degrees of pivoting by limit stops which are not shown. The shell and stud assembly form the gimbal axis of the gyro which indicates by its deviation from the null position the presence of torques resulting from changes in the course.

In the Redstone Arsenal gyro, the spinning wheel and its drive motor which were enclosed in the shell were, at the time our project was started, supported by precision ball bearings. They were replaced in later models by gas bearings of the hydrodynamic type. The increasing trend towards the use of gas bearings in gyros, both for the support of the gimbal and spin axis, was prompted by the requirement of higher and higher accuracy in certain missions of military missiles and, later, spacecraft. To meet tolerances of 0.010 or even 0.001 degree drift rate per hour the center of the spinning mass of a gyro must not be allowed to shift more than about one micro-inch (Reference 1). The friction torque of the gimbal axis of a high performance gyro has to be kept well below one dyne-cm to meet these tolerances. Gas bearings can be designed for very high stiffness, comparable to or better than that of the best ball or roller bearings, i.e. 100,000 pounds per inch or greater. Since the viscosity of gases and vapors is several orders of magnitude lower than that of incompressible lubricants (oils, greases, etc.) friction forces are low. Hydrostatic gas bearings for the support of the gimbal axis of gyros can, therefore, be built with a virtually vanishing pivoting torque and hydrodynamic bearings can operate at very high speeds (250,000 rpm and better) without overheating. Furthermore, gases or vapors are attractive for missions in which difficult environmental conditions are encountered such as extremes of temperature or exposure to nuclear radiation to which gases are relatively insensitive whereas conventional lubricants may become unstable.

Further details on the application and performance of gas bearings may be found in Reference 2.

State of the Art.

I decided, after accepting the project assignment, to first establish the state of the art before starting any analytical or experimental work. Towards the end of 1957, I discussed the problem with the literature search group of the library and, after about six weeks, they came up with a list of almost seventy publications (see Exhibit III).

Not all of these publications were germane to our specific problem and some I could discard without reading them in detail because they pertained to hydrodynamic gas bearings. In the remaining approximately forty papers that I studied thoroughly I noticed a decided lack of satisfactory design data for hydrostatic bearings of the type in which we were interested. Some of them treated the gas as an incompressible fluid. This simplified the analysis considerably; however, the performance of externally pressurized gas bearings is significantly affected by compressibility effects, particularly at higher pressures. No rigorous treatment of the gas as a compressible fluid, subject to isothermal expansion, could be found in any of the articles surveyed.

Single Orifice Pad Bearing

Considering this void in basic information, I came to the conclusion that the multiple orifice type hydrostatic gimbal bearing of the Redstone Arsenal gyro was far too complicated to be treated in a one step approach, analytically as well as experimentally. It became quite clear to me that the behavior of the individual orifice must be understood before the performance of the multiple orifice bearing could be analyzed. I decided, therefore, that we should first study a bearing with a single orifice which can be considered the basic module of more complex multiple orifice configurations and as such the single orifice circular pad bearing of Exhibit IV seemed well suited.

The purpose of the orifice is to regulate the pressure in the "pad", i.e. the thin film of air (or other gas) between the movable and the fixed part of the bearing, as the load varies. If the load increases, the gap h_o between the two decreases and with it the flow of gas because of the greater resistance of the decreased gap. This results in a reduced pressure drop across the orifice and, since the plenum and exhaust pressures are constant, in an increased pressure in the pad, until the load and support forces balance. The process is, of course, reversed if the load decreases.

The gas in the circular pad bearing enters the gap centrally through the orifice and flows radially out to the periphery. It can be shown that the flow in the gap proper is in the laminar regime and that the pressure profile on which the load carrying capacity depends can be calculated from the Navier-Stokes equation for steady, one-dimensional viscous flow. The inertial forces are normally so small, within the range of pressures in which we were interested in this configuration, that they can be safely neglected.

On this basis, I did not find it too difficult to derive the equations for the pressure profile, gap height and flow rate which are given in Reference 3. However, when I attempted to integrate the pressure over the area of the pad to arrive at the load carrying capacity, I ran into considerable difficulties. For weeks, I struggled with the problem developing the expression for the pressure distribution into a

binominal series and carrying the integration to the fifth power term of the series which is poorly convergent. Finally, I had the sense to consult one of the mathematicians at JPL, Dr. Saul Golomb, who showed me that a closed solution for the integral can be found by one of those clever substitutions in which the mathematicians excel. The load carrying capacity could thus be expressed in terms of the error function Erf which can be found in tabulated form in mathematical handbooks. It is useful for the design of orifice regulated bearings to plot the results in normalized form, i.e. the load carrying capacity per square inch of bearing surface. This is shown in Exhibit V in which a family of "lift per unit area" graphs is plotted, with the ratio of the radii of the recess R_o and the periphery R of the circular pad bearing as parameter. All other bearing characteristics of interest, such as the rate of gas consumption, gap height and stiffness could now be calculated as a function of the bearing load.

Tests

The development of the viscous flow theory of the single orifice bearing took the major part of 1958. In the meantime, a test fixture to verify the theory had been designed by Mr. C. Valencia, a test engineer who was assigned to me by Mr. Merrick. It was manufactured by the precision machine shop of the laboratory and consisted of a flexible arrangement for experiments on various configurations of bearing models (see Exhibit VI). The load could be regulated by varying the air pressure on a piston which was attached to the movable bearing element, the "load" plate. The height of the gap h_o which separates the load plate from the (interchangeable) orifice plate was measured electrically by a capacity method. Gas flow rates were determined by variable orifice type flowmeters. Details of the design and experiments are described in Reference 3. The correlation between the experiments and the theory is generally very good. An example is shown in Exhibit VII in which the gap height measurements on a circular pad bearing are represented by the circles and asterisks and the theoretical results by the solid lines.

I was very gratified by these results since they proved that the viscous flow theory described the performance of the single orifice quite accurately, at least within the range of design and fluid parameters in which we were interested. It appeared that it provided a reliable tool to analyze their effect on the performance characteristics of orifice regulated gas bearings.

Examination of the Findings

The gas consumption and the stiffness of a gas bearing are of particular interest. It is obviously desirable to keep the gas consumption of a bearing to a minimum, for reasons of economy and in the interest of low pumping power. It has also been shown (Reference 4) that the

flow should be restricted to stay within the laminar region in the bearing gap and to avoid shock waves and negative pressure profiles with attendant impairment or complete loss of bearing load carrying capacity. Hence, orifices with a small diameter of a few mils* are often used. The lower limit of the orifice opening is determined by mechanical considerations, such as the size of dust particles which may be carried along with the gas stream and must be prevented from clogging the orifices. Low gas consumption is evidently of particular importance in gas bearing applications in spacecraft and missiles because of the severe limitations in power, weight and space. A simple calculation shows, for instance, that on lunar or planetary missions a tank several times the size of the spacecraft would be needed to carry the gas supply for the gyros of the Redstone Arsenal type. An alternative scheme of re-circulating the gas in a closed system is, of course, possible, but requires a compressor with objectionable loading of the spacecraft power supply.

Another important characteristic of a bearing is its stiffness, sometimes also called spring rate, i.e. the degree of displacement of the movable member under load. Most applications require high stiffness. This is especially true of gimbal axis bearings of gyros in which the mass shift under accelerating forces must be held to an absolute minimum. The ideal bearing would be one with infinite stiffness, i.e. the gap would remain constant irrespective of the load.

The viscous flow theory shows that the height of the gap h_o varies with the 1/3 power of the flowrate Q divided by $(P_o^2-P_1^2)$, where P_o is the absolute pressure of the gas at the entrance and P_1 at the exit of the bearing gap. It also reveals (see Reference 3) that the flowrate Q increases with the pressure applied across the orifice which is the difference between the plenum pressure P_{p1}, and P_o, until the ratio P_o/P_1 decreases to the critical ratio which is 0.528 in the case of air. The velocity of the gas then becomes sonic and does not increase further when the pressure is raised: the orifice operates then in the choked condition. This performance is illustrated in Exhibit VIII in which the results of measurements of the air flow Q_{fi} on a pad bearing with a fixed orifice of 3.5 mil diameter are plotted. It is evident that an orifice with a fixed diameter is not conducive to high stiffness since the flowrate is constant below the critical pressure ratio and decreases above it.** As we have seen, it should increase with increasing pressure P_o, i.e. with decreasing $P_{p1}-P_o$ or remain at least constant over as wide a pressure range as possible. Ideally, it should be proportional to $P_o^2-P_1^2$ in which case equation (9) in Reference 3 yields a constant gap height h_o.

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* One mil equals .001 inch.

** The abscissa of Exhibit VIII is not actually the pressure ratio P_o/P_{p1} but only the bearing entrance pressure P_o; the plenum pressure is constant at P_{p1} = 40 psig. Variation of orifice diameters with pressure ratio are also plotted.

Manipulation of the Flowrate

The conclusion which I drew from examining the equations, i.e. that the design of a hydrostatic gas bearing with the conventional fixed orifice was evidently not resulting in optimum stiffness, was very disquieting to me. For weeks, I wondered whether anything could be done about it, and if so, would it be practical and not prohibitive from the standpoint of production and cost. To improve the stiffness, the flowrate had to be "manipulated" to increase with increasing load, as we have seen. This is contrary to the performance of a fixed orifice operating from a plenum with constant supply pressure. A possible solution occurred to me one day when I was again brooding over the problem. The rate of flow through an orifice with a fixed opening is a function of the supply pressure P_{pl} as well as the load. Why not regulate the supply pressure as the load varies and thereby adjust the flowrate to approximate as closely as possible the ideal characteristic? This, however, simple as it was to do manually, proved a very difficult task to implement automatically. I finally discarded the idea as being too complicated and impractical, particularly for aerospace applications.

I went back after that to the equations in Reference 3 and took another good look at equation (6), which shows that the flowrate Q is proportional to the area A_2 of the orifice opening:

$$Q = C_D \phi A_2 \left[2\rho_{pl}(P_{pl} - P_o)\right]^{1/2}$$

where

$$\phi = \left[r^{2/k} \left(\frac{k}{k - 1}\right) \left(\frac{1 - r^{\frac{k-1}{k}}}{1 - r}\right) \left(\frac{1 - \left(\frac{A_2}{A_1}\right)^2}{1 - \left(\frac{A_2}{A_1}\right)^2 r^{2/k}}\right) \right]^{1/2}$$

and

C_D = orifice coefficient.

A_1 = area of upstream flow passage.

A_2 = area of orifice opening.

P_{pl} = absolute pressure of gas upstream, in plenum.

P_o = absolute pressure of gas past orifice, at entrance to bearing gap.

ρ_{pl} = density of gas in plenum.

r = P_o/P_{pl}

k = ratio of specific heats of the gas, c_p/c_v.

The thought occurred to me that there was nothing sacrosanct about an orifice with a fixed opening A_2 although in the past it had been standard in the design of all orifice regulated gas bearings known to me. If a way could be found to modulate the area of the opening in the right direction as the load varied an improvement of the stiffness and gas consumption could be expected. The ideal of infinite stiffness could be approached if the flowrate could be made as closely as possible proportional to $P_o^2 - P_1^2$. I gave, therefore, some thought to how the orifice area could be made to "follow" load changes automatically. One possible answer was to use an iris type orifice whose opening would be controlled by the output of a strain-gage which measured the loading force. I discussed this concept with Mr. Frank Batsch, a very competent design engineer and graduate of Rice University. He had joined my group some time before because he was very intrigued by the gas bearing project after he had previously been assigned to it part-time by the design and engineering section of which he had been a member. Frank Batsch made several sketches of an iris type orifice whose sliding segments could be moved and whose opening thus be controlled by the output of a sensor of the loading force in response to changes in the load. We came, however, to the conclusion that such a device was not reliable enough because of problems of friction and stiction and was too complicated and therefore impractical, in particular for a spacecraft gyro with not one, but a large number of orifices.

The Elastic Orifice

The thought occurred to me then that it might be easier to control the orifice area by the gas pressure in the bearing instead of the load forces. In particular, the differential pressure $P_{p1}-P_o$ across the orifice seemed a "natural" to accomplish this. Frank Batsch agreed and went to work with the implementation of this idea. He came up with two alternative suggestions. The first one, a metallic flexible orifice, is shown in Figure 6 of Exhibit IX. It consists of a metallic membrane with a conical plug attached to it which moves in an opening of a slightly different cone angle. The annular opening between the two through which the gas enters from the plenum varies in size with the deflection of the membrane under the influence of the pressure drop across it. This design had the advantage that it could be expected to operate satisfactorily under extreme environmental conditions, as they are encountered in space, including extremes of temperature and nuclear radiation. However, it had the disadvantage that the membrane area had to be pretty large to produce sufficient deflection and that it was fairly expensive to manufacture.

The second suggestion of Frank Batsch was to use as an orifice a circular disk of silicone rubber or some other elastomer, and to control the size of an opening through the elastic deformation produced by the pressure forces. Several modifications of this concept are illustrated in Figure 3 of Exhibit IX. This approach appealed very much to me and I authorized Frank Batsch to proceed with its implementation.

He contacted several manufacturers of elastomers and was very encouraged when he learned how easy it was to mold the elastic orifices from commercially available silicone rubber liquids. The circular disk with a small central hole, shown in Figure 3a in Exhibit IX, was the simplest form to start with. Frank Batsch designed a two-part mold which is shown schematically in Figure 1 of Exhibit IX with a central pin of 3.5 mil diameter. He coated it with a mold-release and around it he poured the elastomer, a viscous fluid prepared in accordance with the manufacturer's instructions. After curing, the pin can be easily withdrawn, leaving the elastomer disk with its central hole. It is then removed from the mold and cemented in place in the orifice plate. The manufacturing process is quite simple and inexpensive. It lends itself to production in larger quantities, for multiple orifice gas bearings, etc., by means of multiple molds.

Experiments

Next, we decided to test the performance of an elastic orifice and compare it with the characteristics of a fixed orifice which we had obtained in previous tests. We replaced the fixed orifice in the orifice plate of the test fixture shown in Exhibit VI with a silicone rubber orifice with the same diameter of 3.5 mil of its central opening in its relaxed condition, i.e. without pressure applied. We held the downstream pressure constant by venting the orifice to the atmosphere and varied the upstream pressure between 8 and 48 psig. The flowrate Q was measured with a variable orifice type flowmeter (Rotameter).

The results of this test are shown in Figure A-6 of Exhibit IX.

The flowrate Q_{el} of the elastic orifice decreases with increasing pressure, whereas the flowrate Q_{fi} of the fixed orifice shows the opposite behavior. The diameter D of the orifice opening can be calculated from the measured flowrate and pressure by using the equations on page 7 . The effective diameter of this elastic orifice shrinks from 3.5 mil in the relaxed, undeformed condition to 2.5 mil when a differential pressure of 48 psig is applied across it. This corresponds to a shrinkage of the orifice opening area A_2 of approximately 2 to 1.

The rate of shrinkage can be modified by varying the diameter and thickness of the elastic orifice. A wide range of gas flow and stiffness characteristics of a gas bearing can thus be obtained. This has been verified in a later investigation which the Stanford Research Institute carried out under contract for NASA (see Exhibit IX, Figures 4 and 5). Figure 5 of this Exhibit shows that the ideal of a bearing with infinite stiffness can be closely approximated by using an elastic orifice.

In order to study the effect of elastic orifices on the performance of some typical gas bearings we built two experimental single orifice

bearings and compared their behavior with elastic versus fixed orifices. One was a circular pad bearing which was operated at 40 psig plenum pressure, first with a fixed orifice of 3.5 mil opening diameter, then with an elastic orifice with the same diameter in the relaxed condition. The results are plotted in Exhibit VIII, which shows that the gas flow Q_{el} through the elastic orifice is, over most of the operating range, considerably lower than that of the fixed orifice, Q_{fi}. If the fixed orifice is operated in the choked condition, which is common bearing practice, it consumes approximately 2.6 times as much gas as the elastic orifice.

The second experimental model was a spherical bearing illustrated in Figure 7 of Exhibit IX. When we equipped this bearing with an elastic orifice, its gas consumption Q_{el} dropped to 31.5% of Q_{fi}, the gas flow with a fixed orifice (see Figure 5 in Reference 5). At the same time, the stiffness of the bearing is improved substantially over a wide range of loads, as can be seen from the slope of the $h_o = f(w)$ curves in Figure 5 of Reference 5.

We were very gratified with these results and felt that we had now demonstrated the advantages of elastic orifices in gas bearings sufficiently. The saving in gas consumption, for instance, which resulted from the replacement of a fixed orifice with an elastic orifice is particularly significant for gas lubricated instruments on board space vehicles which can thus travel about three times further with the same initial supply of gas.

We felt, however, that it was important to study the deformation of elastic orifices under pressure more closely, to gain a better understanding of the mechanism and optimize the design. The elastic deformation depends on the applied pressure and the dimensions, modulus of elasticity and Poisson's ratio of the elastomer. I suggested to Mr. Houston McGinness, a specialist in applied mechanics who was a member of my section, that he treat the problem analytically. He did so by applying equations from the small-deformation theory of elasticity to a model of the circular elastic orifice with a central opening. The results are published in Reference 6. In order to verify them experimentally, we built a large scale transparent model of an elastic orifice on which we could study the elastic deformation by photography and measurement more accurately than on the small size orifices of our previous experiments. The model and tests are described in detail in Reference 7. The correlation with the theoretical treatment by Mr. McGinness is satisfactory, considering the assumptions which had to be made. The error ranges from a minimum of 6% to a maximum of 28%.

Applications of Elastic Orifices

By this time, JPL was no longer involved in work on military missiles and I do not know whether elastic orifices ever found application in missiles. For spacecraft which cruise for long periods of time,

celestial guidance was found to be more practical than gyros which require a supply of pressurized gas for the bearings. There has been widespread interest, both within JPL and outside, in other applications however. I know, for example, that Boeing and other companies used the published results of our research in the design of spherical gas bearings for large attitude control simulators.

Our experiments with different configurations of elastic orifices which are described in Reference 7 showed that they could be designed to close the opening and shut off the flow of gas completely on applying the proper pressure. This gave me the idea that the principle could be used for pneumatic logic elements in computing and controlling systems. While fluid logic elements are considerably slower than their electronic counterparts, they offer a number of advantages which are especially important in space applications because they are less sensitive to nuclear radiation and extremes of temperature and are not subject to electromagnetic interference.

A number of such logic elements were designed by Frank Batsch and showed satisfactory performance on test, including a speed of response of approximately 10 milliseconds. Reference 8 describes the tests and potential applications of elastic orifices as logic elements in greater depth.

NASA, under whose cognizance JPL came several years ago, has issued NASA TECH BRIEF 63-10123, "Elastic Orifice Automatically Regulates Gas Bearings," in June 1964, to explain the operation of elastic orifices in gas bearings and to stimulate interest in their application. As a result of this publication NASA has received many inquiries for further information from industrial organizations, expressing considerable interest in this development; to date, however, no specific case of pratical application of elastic orifices has been reported. The recent release of the more comprehensive NASA Report SP-5029 (Exhibit IX) may stimulate further interest in this device.

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EXHIBITS

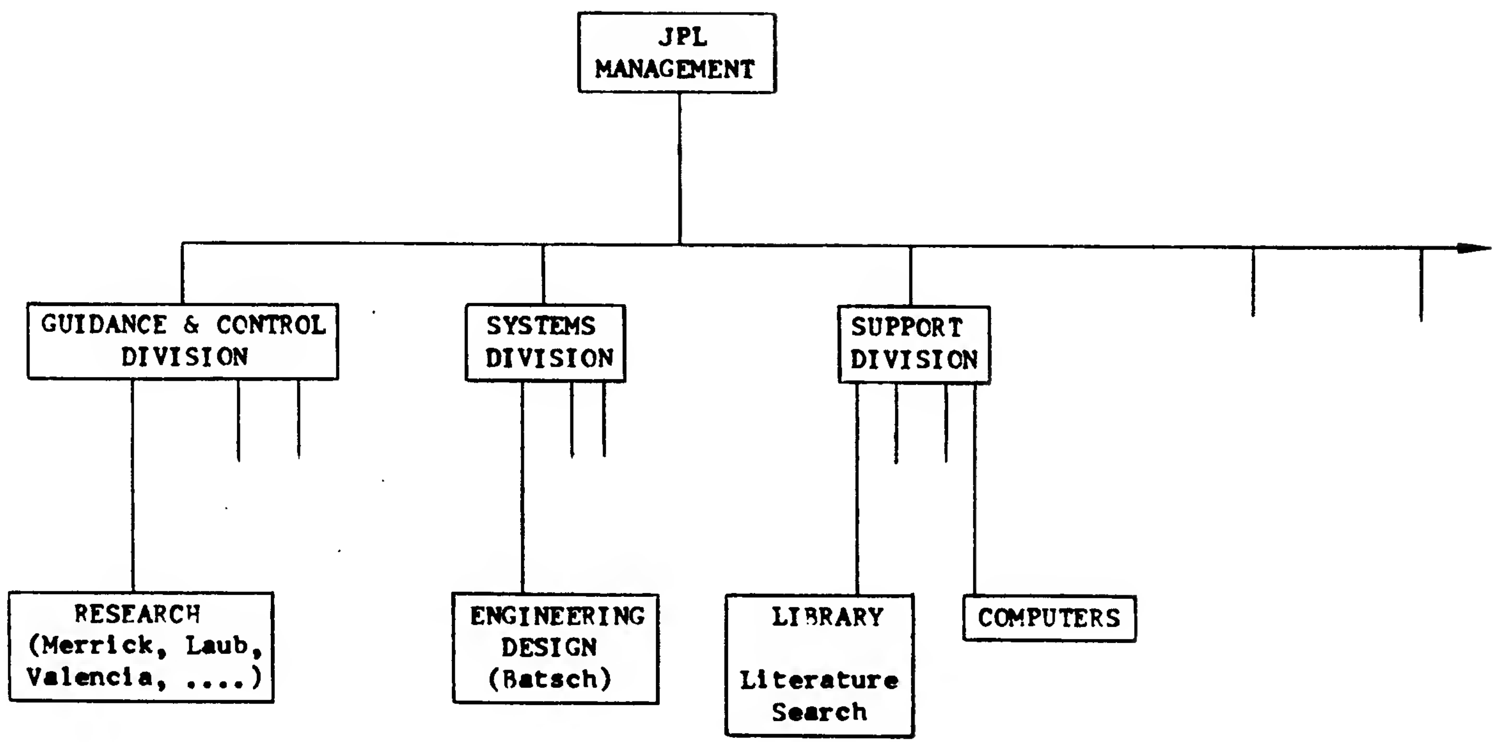

Exhibit I: Partial Organization Chart of JPL in 1957.

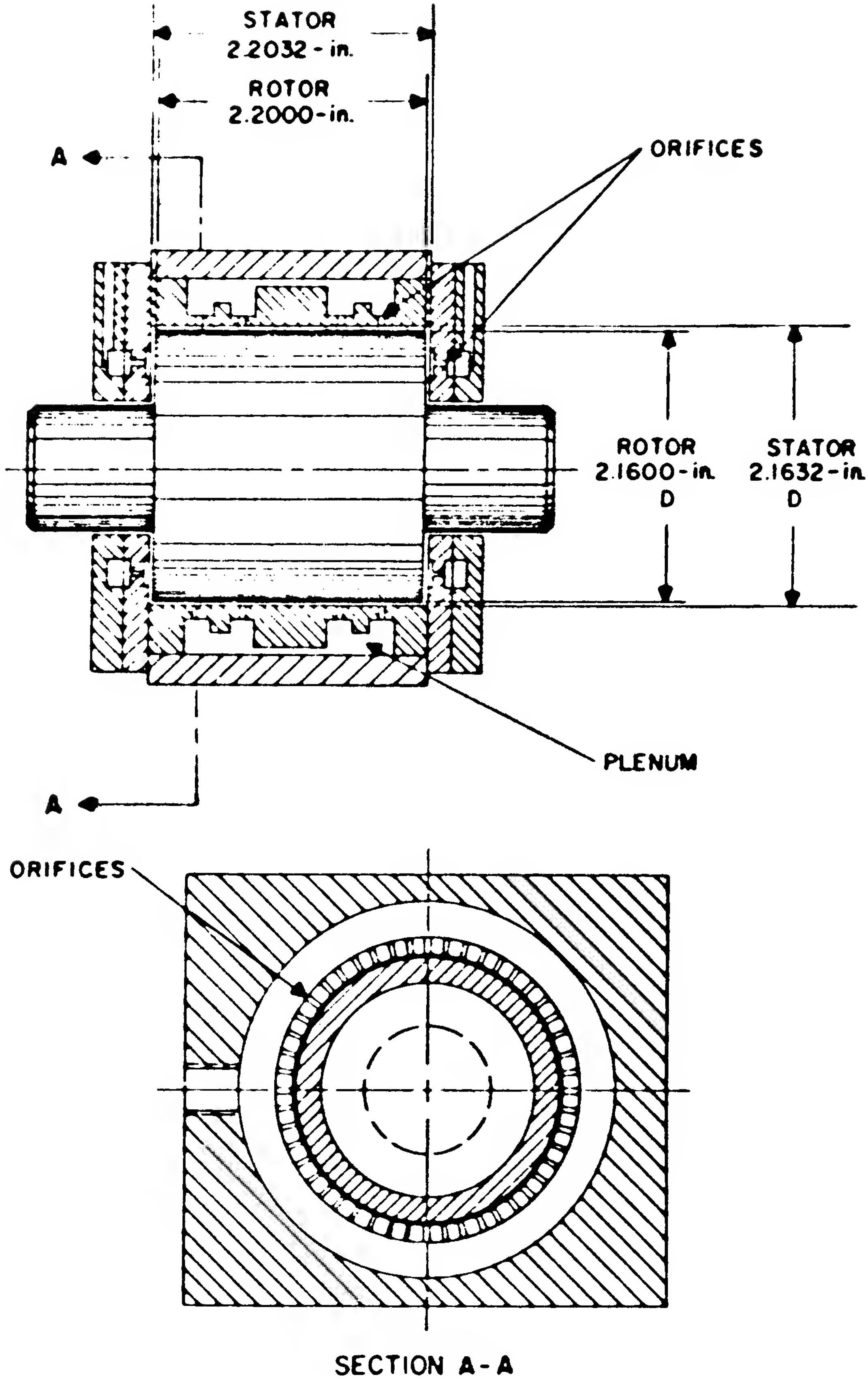

Exhibit II: Redstone Arsenal Gyro.

Exhibit III: Gas Bearing Literature

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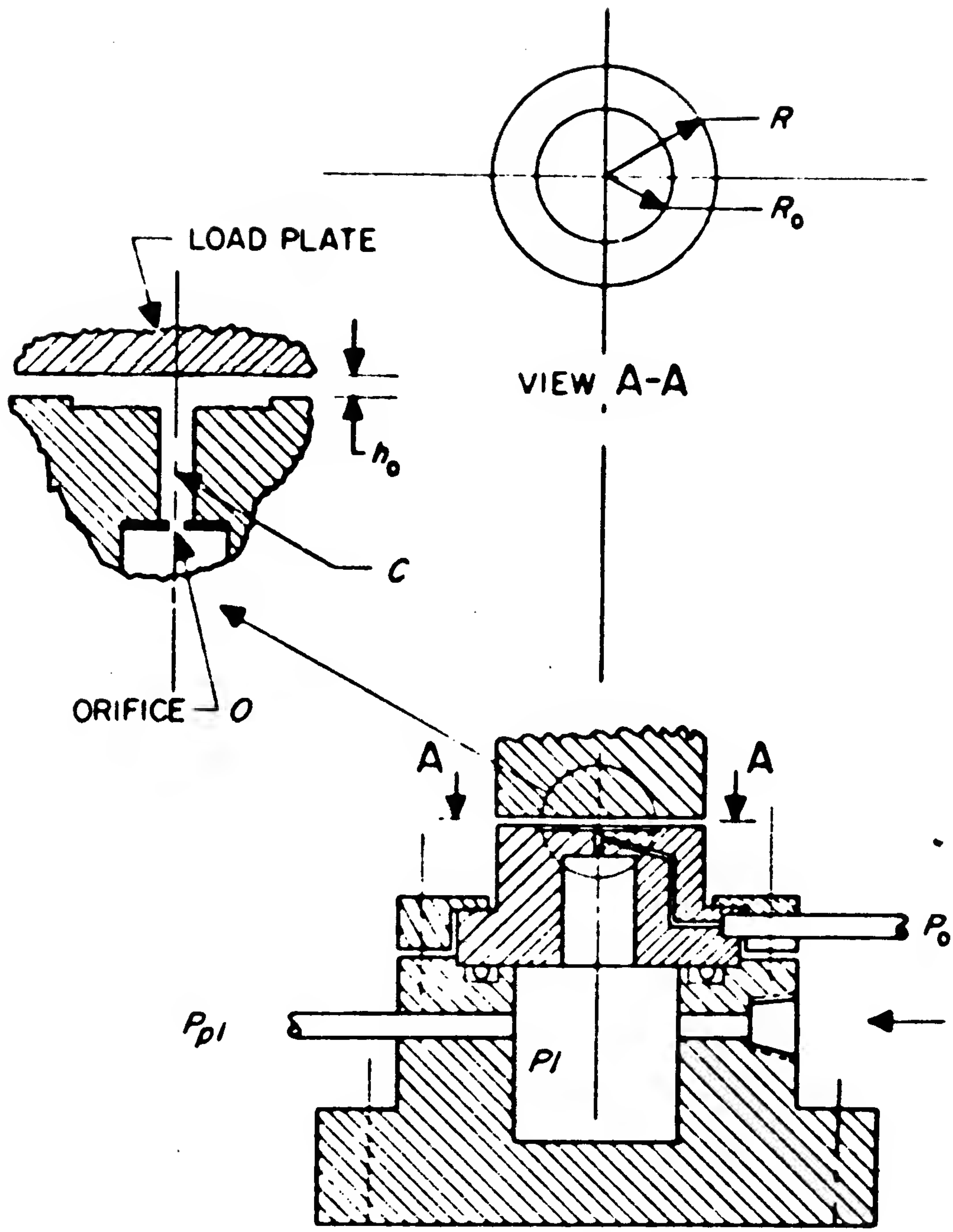
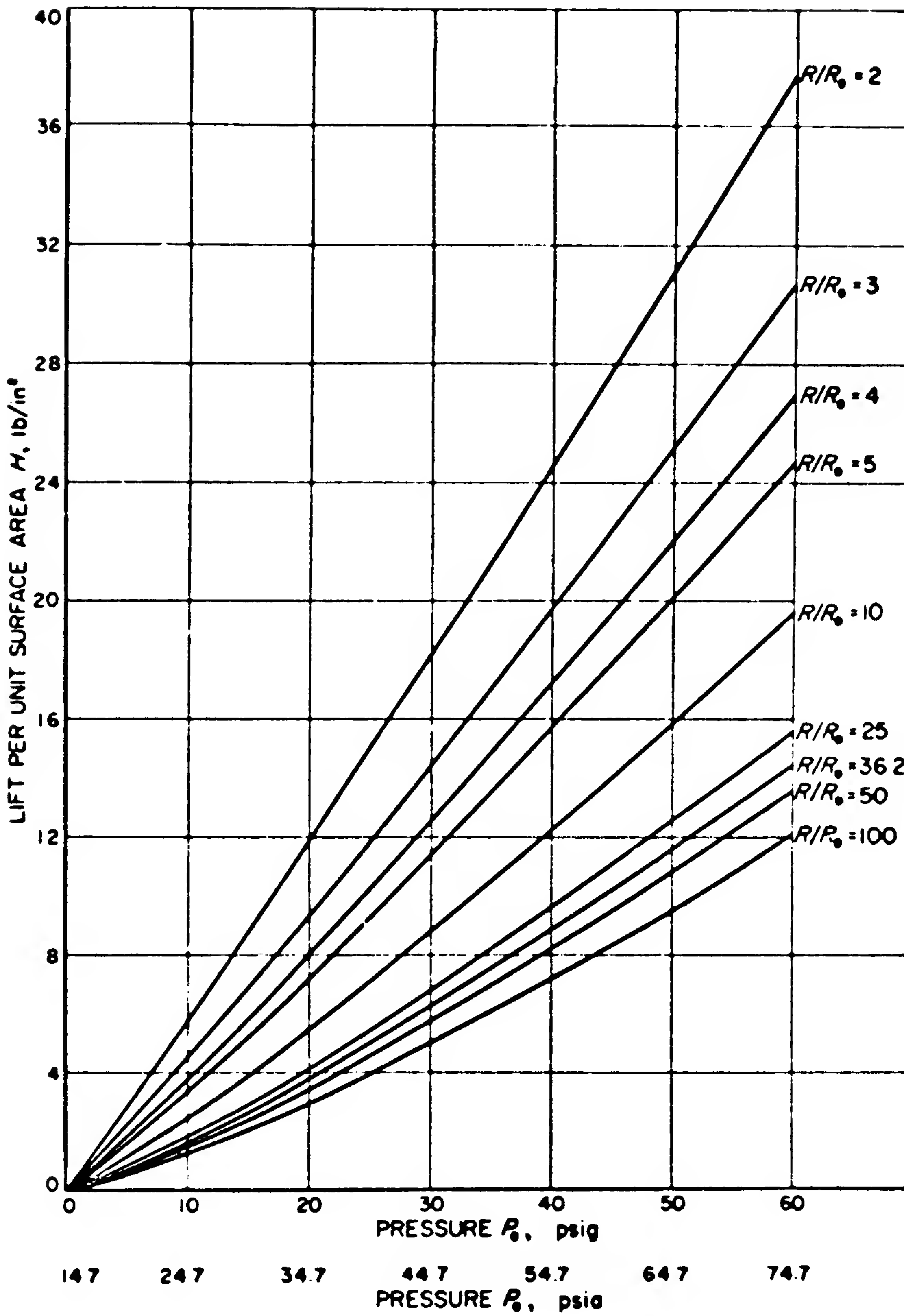
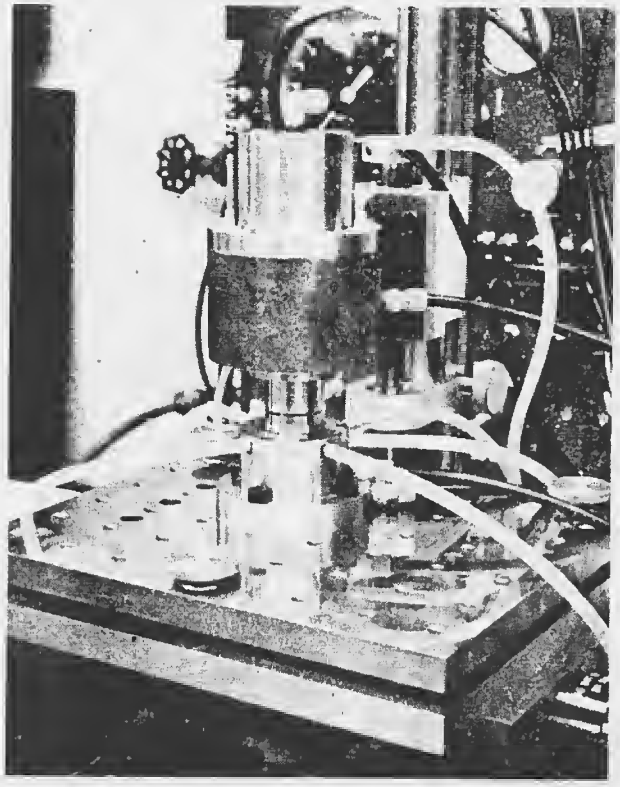


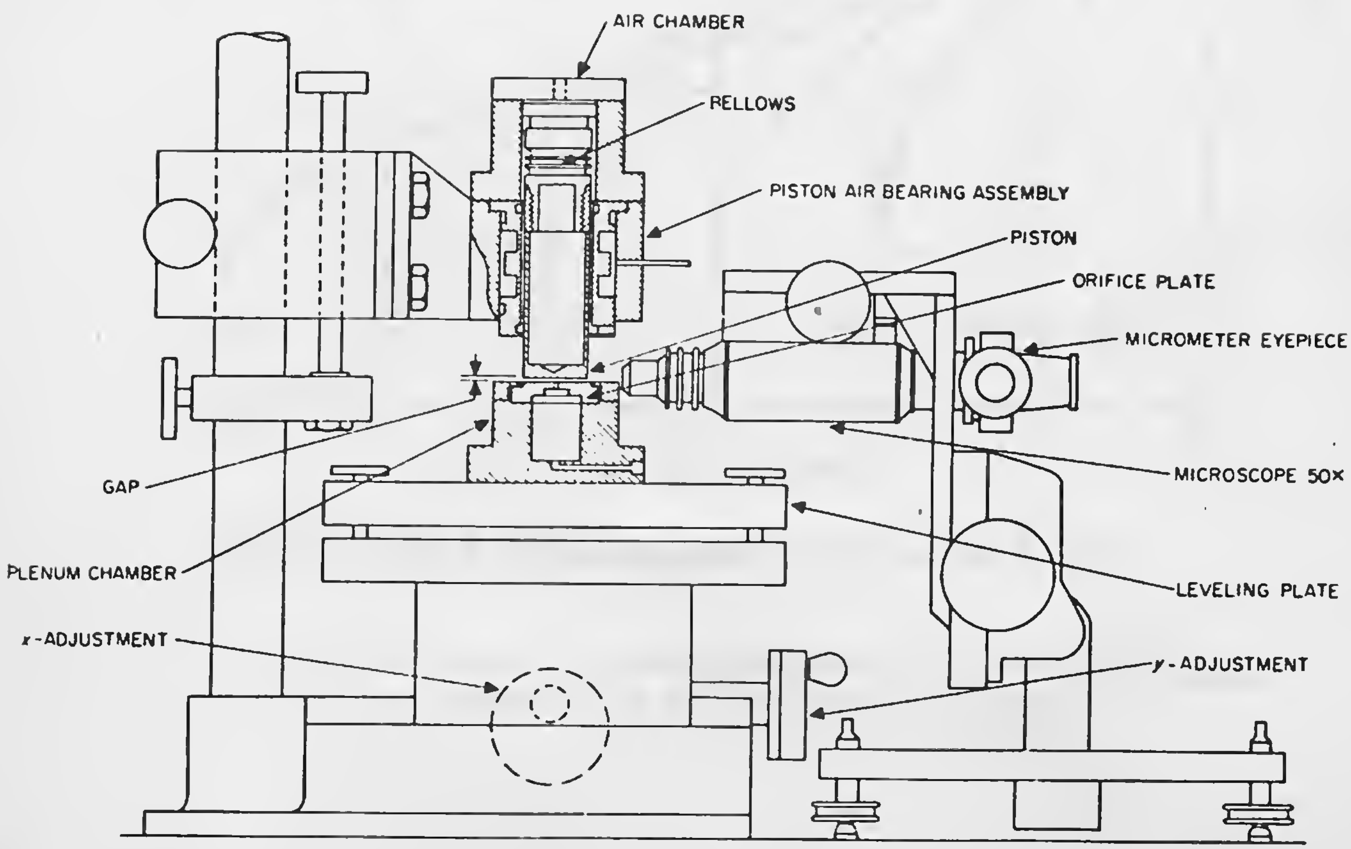
h_o - gap height.
R - radius of circular bearing pad.
R_o - radius of recess in bearing pad.
P_o - pressure of gas past orifice, at entrance to bearing gap.
P_{p1}- pressure of gas in plenum, upstream of bearing.

<u>Exhibit</u> <u>IV:</u> Single Orifice Bearing.

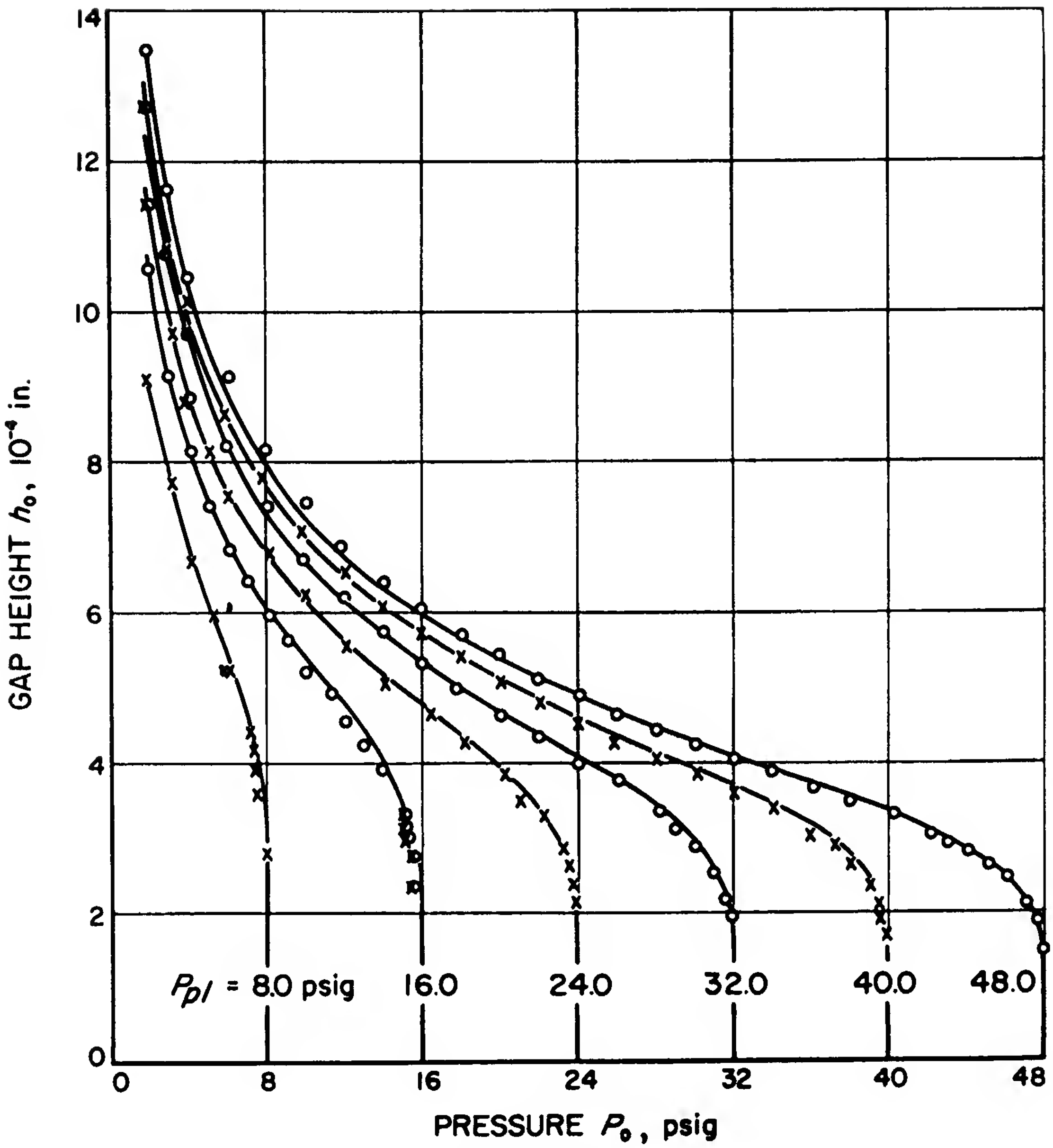


<u>Exhibit V:</u> Load Carrying Capacity of Single Orifice Bearings. Symbols Same as Exhibit IV.



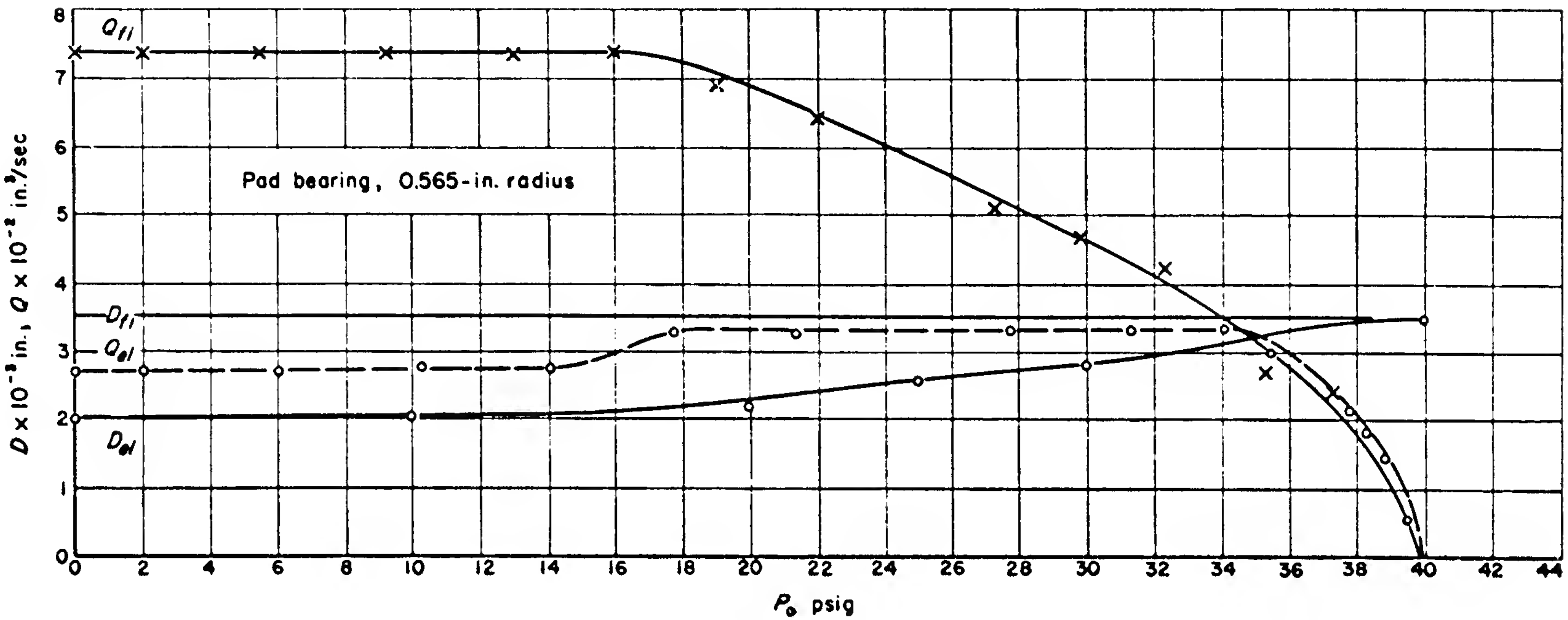


<u>Exhibit</u> <u>VI:</u> Test Fixture.



Pad bearing gap measurements: $R/R_0 = 36.2$.

__Exhibit VII:__ Bearing Pad Gap Measurements. Curves are Lines of Constant Plenum Pressure P_{pl}. Symbols Same as Exhibit IV.



Performance of fixed and elastic orifices in pad bearing

P_o = pressure of gas past orifice, at entrance to bearing gap.
Q_{fi} = air flow rate through fixed orifice.
Q_{el} = air flow rate through elastic orifice.
D_{fi} = diamter of fixed orifice.
D_{el} = diameter of elastic orifice.

<u>Exhibit</u> <u>VIII:</u> Flow Rate Measurements. Plenum Pressure P_{pl} Held Constant at 40 psig.